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J. Polman

A. K. de Jonge

A. Castelijns

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J. Polman, A.K. de Jonge and A. Castelijns
Philips Research Laboratories
Eindhoven - Netherlands

1. INTRODUCTION

A number of papers has already been published on electrodynamic oscillating compressors. Particular attention has been given to oscillating compressors equipped with a metal spring for tuning the mass-spring system to the operating frequency of the compressor [1-4]. The advantages of such compressors as compared with those driven by rotary electric motors are related to the smaller number of moving parts and bearings required and to lower side forces on the piston [3].

Less attention has been given to the free piston electrodynamic compressor, although some work in this respect has been carried out [5,6]. One advantage of a free piston system is that no expensive and complicated spring construction is required, the compressed gas being used as a spring.

An interesting property of the oscillating compressor is that its output can be controlled by adjusting the stroke of the pistons, being dependent on the applied voltage. This is in contrast to a rotary motor driven compressor of which the output can be controlled by varying the speed of the motor, which is often more complicated than varying the applied voltage. Controllability of the compressor is advantageous for the performance of a total system, particularly for that of a heat pump system.

The purpose of the present paper is to give some data on a small experimental (200 W) double acting free piston compressor and some experimental data on its performance. The gas used in the investigations was nitrogen.

Section 2 of this paper describes the design of the compressor, including its linear motor, section 3 gives some information on the motion and the stability of the free piston, while results of measurements on the performance are presented in section 4. Finally, section 5 gives some conclusions of the present work.

2. DESIGN OF THE COMPRESSOR

2.1. General set-up

A schematic set-up of the compressor is given in figure 1. The moving part of the compressor is equipped with two balanced coils rather than one which has some advantages as far as demagnetization and iron losses are concerned [4]. The coils are connected to the 50 Hz voltage supply by means of flexible wires. The permanent magnet is a ferroxdure-300 loudspeaker magnet. The maximum stroke and the diameter of the cylinders of the compressor were taken 0.02 m. The valves were taken from a normal refrigerating compressor and are of the reed-valve type. More design parameters are given in section 2.2.

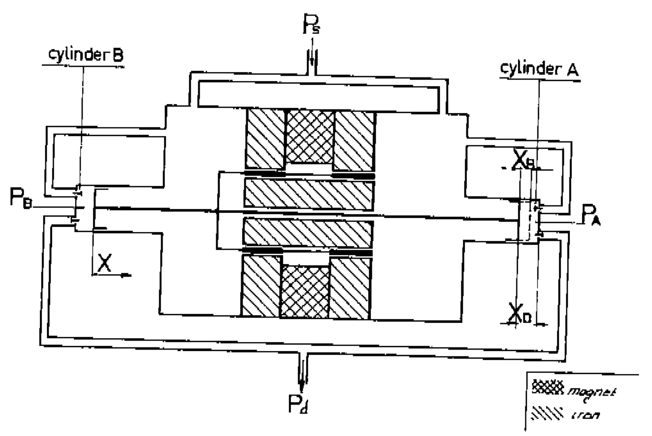

Fig.1. Schematic set-up of the compressor.

2.2. Optimization of the oscillating compressor

The equation of motion of the piston system, including coils and connecting structure (see figure 1), can be written as

$$m\frac{d^2x(t)}{dt^2} + S(p_A(t)-p_B(t)) = B_g l(x)i(t) \quad (2.1)$$

where m, S, x, B_g and i are the mass of the moving part of the compressor, the piston surface, the piston position, the magnetic flux density in the gaps between pole shoes and iron centre and the current, respectively. p_A and p_B are the pressures in both cylinders and $l(x)$ is the length of that part of the wires of the two coils in the gaps, as determined by the geometry of the compressor. Outside the gaps the magnetic flux density is taken zero. Equation (2.1) does not take account of forces by friction and iron losses.

The equation describing the electric circuit is

$$U = U_{ind} + iR + L\frac{di}{dt} \quad (2.2)$$

where $U = U_o \sin\omega t$, the externally applied voltage, $U_{ind} = B_g l(x)\frac{dx}{dt}$, the voltage induced by the motion of the coils in the magnetic flux, R is the resistance of the coils and L the inductance, taken independent of coil position. With the following substitutions equations (2.1) and (2.2) can be made dimensionless: $\xi = x/x_d, \varphi = \omega t, \omega_r^2 = S(p_d-p_s)/mx_d$, where x_d is the distance between the midposition of the pistons and the cylinderheads (see figure 1) and where p_d and p_s are the discharge and suction pressure of the compressor,

$\hat{p}_A = p_A/(p_d-p_s)$, $\hat{p}_B = p_B/(p_d-p_s)$, $\hat{l}(\xi) = l(x)/l_s$
where l_s is the total length of the wire of the coils,

$$\hat{i} = i/i_o \text{ with } i_o = B_g l_s \omega x_d/R,$$

$C = B_g i_o l_s/(m\omega^2 x_d)$, $\tau = \omega L/R$, $\hat{U} = U_o/(B_g l_s \omega x_d)$.

The equations obtained are

$$\frac{d^2\xi}{d\varphi^2} + \left(\frac{\omega_r}{\omega}\right)^2(\hat{p}_A-\hat{p}_B) = C\hat{i}\hat{l}(\xi) \quad (2.3)$$

$$\tau\frac{d\hat{i}}{d\varphi} + \hat{i} = \hat{U}\sin\varphi - \hat{l}(\xi)\frac{d\xi}{d\varphi} \quad (2.4)$$

These equations have been solved by means of a Runge-Kutta method for various values of $\omega_r/\omega, \tau$, $\hat{U}$, p_s, p_d and C assuming an adiabatic compression process for an ideal gas, without valve losses. The boundary conditions are given by the requirement of periodicity of the solutions. It has been shown that the results of the calculations are hardly sensitive to changes in the ratio of the magnet pole shoe length to the length of the coils when it is approximately one, and so, in the following, this ratio is taken constant and equal to one, as illustrated in figure 1.

Figure 2 shows, for p_d = 8 bar, p_s = 1 bar τ = 0.4, and C = 2, the effect of the reduced voltage $\hat{U}$ and (ω_r/ω) on the electrical efficiency of the compressor, defined by

$$\eta_e = 1 - \langle i^2R\rangle/P_{in} \quad (2.5)$$

where P_{in} is the output power. It appears that the resonance is not very pronounced, particularly at higher values of $\hat{U}$, which is due to the strong damping of the system. It has been shown that C = 2 is a reasonable value in terms of sensitivity to changes in C and a maximum of η_e for an acceptable normalized piston amplitude ξ_b, being a measure for the output flow.

Figure 3 and 4 show the effect of the discharge pressure on the performance of the compressor for various values of $\hat{U}$ and for $(\omega_r/\omega)^2 = 0.16 \times (p_d-p_s)$, the value corresponding to the maximum for $\hat{U}$=1.0 in figure 2, and C = 2. Again it is illustrated that ξ_b should exceed the minimum stroke required for exhausting, which is given by

$$\xi_d = ((p_d/p_s)^{1/k}-1)/(p_d/p_s)^{1/k}+1) \quad (2.6)$$

where k is the adiabatic constant.

Finally figure 5 shows the output flow given by the product of the volumetric efficiency η_v and amplitude ξ_b versus pressure ratio with the same parameters $(\omega_r/\omega)^2$ and C as in figures 3 and 4. This figure illustrates the controllability of the oscillating compressor: by changing the voltage the output can be varied in dependence of the pressure ratio required.

From the above calculations it was concluded that for a compressor optimized for p_d = 8 bar and p_s = 1 bar the value to be taken for $(\omega_r/\omega)^2$ is 1.12 and the value for C is 2. This means that the following design data have been obtained:
x_d = 0.01 m, S = 31.4 10^{-3} m^2, m = 0.20kg, B_g = 0.57 T, R = 40Ω, l_s = 125 m, $U_o/\hat{U}$ = 224 V.

3. MOTION AND STABILITY OF THE FREE PISTON

A problem that is peculiar to a free piston device is that of the stability of the mid position of the pistons. Since the pistons are not connected to the housing of the compressor, their position is determined only by their motion and the gas forces. Drift of the moving part of the compressor may therefore occur, resulting in a collision between piston and cylinder head. The fact that a free piston compressor is unstable with respect to such a drift can be illustrated in the following way.

Simplifying the equations 2.1 and 2.2 describing the behaviour of the compressor as given in section 2 by taking l(x) independent of x, introducing a friction term $c_f\frac{dx}{dt}$, and relinguishing the condition

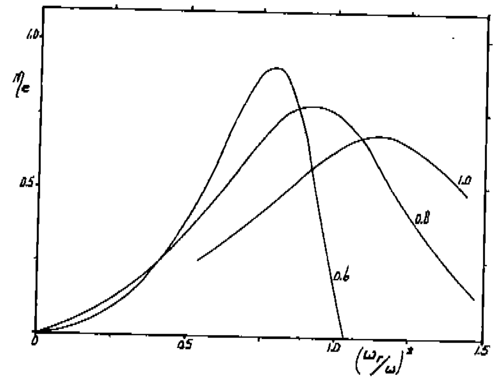

Fig. 2. Electrical efficiency η_e versus $(\omega_r/\omega)^2$ for various values of $\hat{U}$. p_d=8 bar, p_s=1 bar, C=2, τ=0.4

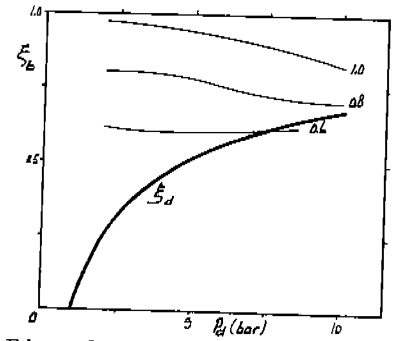

Fig. 3. Normalized amplitude ξ_b versus p_d for various values of $\hat{U}$. ξ_d is defined in (2.6). p_s=1 bar, C=2, τ=0.4. $(\omega_r/\omega)^2$=0.16(p_d-p_s)

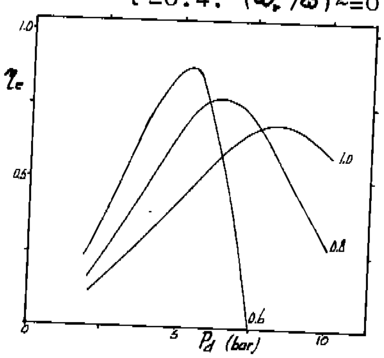

Fig. 4. Electrical efficiency η_e versus p_d for various values of $\hat{U}$. The parameters are identical to those of fig. 3.

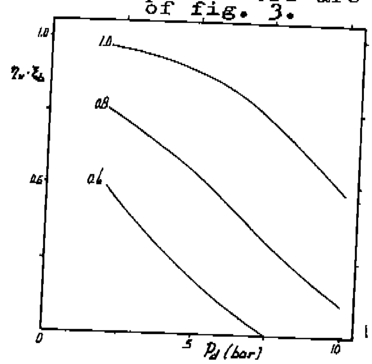

Fig. 5. Normalized output $\eta_v \cdot \xi_b$ versus p_d for various values of $\hat{U}$. The parameters are identical to those of fig. 3.

of periodicity of the solutions. One obtaines an equation for the mid position x_m of the piston system, that can be solved semi-analytically.

Drift of the piston system will clearly occur if the solution for $x_m(t)$ is unstable, i.e.

$$\frac{dxm}{dt} > 0 \text{ if } x_m > 0 \qquad (2.7)$$

The drift generating term of the equations (i.e. the second term of equation 2.1) can be calculated numerically for various x_m and proves to be negatively proportional to x_m for small x_m, i.e.

$$\frac{S}{T}\int_0^T (p_A(t)-p_B(t))dt = k_1 x_m \qquad (2.8)$$

where $k_1 < 0$. This means that the solution is unstable and that drift may occur. The equation shows that the drift motion is proportional to

$$|k_1|/(1+\frac{(B_g l_s)^2}{C_f R}), \text{ and that the}$$

drift decreases with increasing $B_g l_s$, due to the induced voltage.

Analysis of the constant k_1 for the present compressor and comparison with the quantity $k_2 = (p_d-p_s)S/x_d$, representative for the spring rate in a linearized description of the compressor, shows that $|k_1| < k_2$. This means that it is possible to stabilize the drift of the pistons with a metal spring having a stiffness k_s with $|k_1| < k_s < k_2$: Introduction of this spring does not affect the resonance frequency of the compressor, since $k_s < k_2$, but stabilizes the drift since $k_s > |k_1|$.

The motion of the pistons has been studied experimentally by analysing the pictures obtained using a high speed film camera (10000 frames/second). The position of the pistons has been Fourier-analysed. The Fourier components for a typical case are given in table 1. From this table it can be seen that the motion of the piston can be described accurately by taking the first harmonic only.

TABLE 1

a_0	0.188		
a_1	9.051	b_1	2.320
a_2	0.031	b_2	-0.004
a_3	0.222	b_3	-0.129
a_4	0.001	b_4	-0.015
a_5	0.005	b_5	-0.011

Fourier analysis of piston position

$$x = \sum_{n=0}^{5} (a_n \cos n\omega t + b_n \sin n\omega t)$$

Parameters: p_s = 1.0 bar, p_d = 9.1 bar, $\langle U \rangle$ = 150 V, P_{in} = 160 W.

Since these and similar measurements show that the motion of the pistons is almost sinusoidal it may be concluded that a study of the properties of the compressor part of the device can be made separately from a study of the linear motor part. This is advantageous in both analysing the experimental results and in the computer simulation of the free piston electrodynamic compressor.

4. MEASUREMENTS ON THE COMPRESSOR

The compressor as described schematically in section 2 of this paper was investigated by measuring the electrical input power, pressures, output flow and the PV-diagram. Some typical examples of results are given in figures 6-8. It should be noted that the actual data deviate slightly from those given in section 2. In particular the measured value of B_g was found to be somewhat lower than expected. This has an effect on the actual value of the constant C, but as mentioned in section 2 the effect of such a change is not very pronounced.

Figure 6 gives the electrical efficiency of the compressor for two supply voltages, as a function of discharge pressure with a fixed suction pressure of 1 bar. It is seen that the results agree qualitatively with those of the calculations. Differences may be due to the fact that friction and iron losses have been ignored in the calculations. The total efficiency of the compressor, as given in figure 7 as a function of a discharge pressure p_s = 1 bar, is determined by comparing the output with that of an adiabatic compression process in a compressor with the same geometry and stroke and with no-loss valves and no heat losses. The experimentally determined efficiency equals that of a conventional refrigerating compressor of the same size. Improvement is possible, however, by using better designed valves.

Finally figure 8 gives the output flow of the compressor as a function of discharge pressure for various voltages. This figure illustrates that control of the compressor output is possible by adapting the supply voltage to the required pressure ratio and output.

5. CONCLUSIONS

In this paper it has been shown that a free piston electrodynamic gas compressor can have stable operation without drift of its moving part and with an acceptable adiabatic efficiency. The resonant character of the compressor is less pronounced than that of a metal-spring oscillating compressor. This is an advantage in view of the various operating conditions of the compressor. A compressor of the present

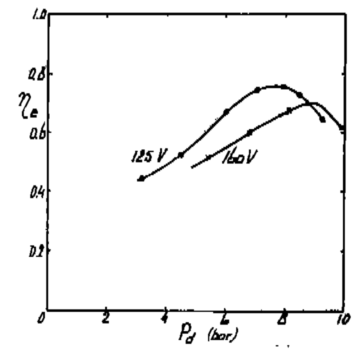

Fig. 6. Electrical efficiency η_e versus p_d for $\langle U \rangle$ = 125V and 160V. p_s=1 bar

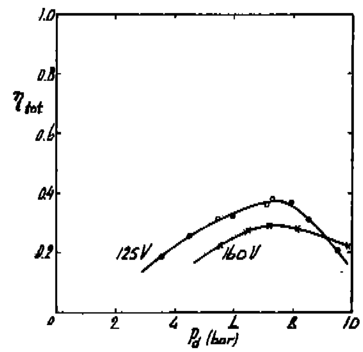

Fig. 7. Total efficiency η_{tot} versus p_d for $\langle U \rangle$ = 125V and 160V. p_s = 1 bar

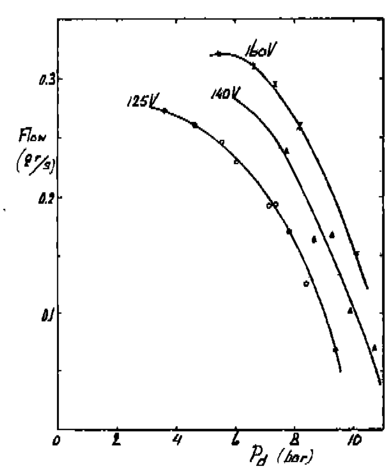

Fig. 8. Output versus p_d for $\langle U \rangle$ = 125, 140 and 160V. p_s = 1 bar.

type may be of interest for use in refrigerating equipment.

The stroke of the oscillating compressor can be controlled by adjusting the applied voltage. By this the output of the compressor can be controlled and adapted to the requirements. This makes it feasible to use the controllable free piston compressor in a heat pump system, in this way improving the seasonal coefficient of performance of such a system considerably.

The efficiency of the present experimental compressor is not sufficiently high for use in a heat pump, but improvement, i.a. by introduction of better designed valves, can be foreseen. Attempts will be made to achieve a higher compressor efficiency in the near future.

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